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RESULTS CF THE EXPERIMENTAL STUDY OF THE EFFECT
ON COMPRESSOR PARAMETERS FROM WATER
ADMITTED AT THE INLET TO A
CENTRIFUGAL COMPRESSOR

by

A. S. Moskalenko, N. L. Zel'des

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М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

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GREEK ALPHABET

Alpha	Α	α	∝	Nu	Ν	ν	
Beta	Β	β		Xi	Ξ	ξ	
Gamma	Γ	γ		Omicron	Ο	ο	
Delta	Δ	δ		Pi	Π	π	
Epsilon	Ε	ε	ϵ	Rho	Ρ	ρ	ϱ
Zeta	Ζ	ζ		Sigma	Σ	σ	ς
Eta	Η	η		Tau	Τ	τ	
Theta	Θ	θ	ϑ	Upsilon	Τ	υ	
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Kappa	Κ	ϰ	κ ϰ	Chi	Χ	χ	
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Mu	Μ	μ		Omega	Ω	ω	

RUSSIAN AND ENGLISH TRIGONOMETRIC FUNCTIONS

Russian	English
sin	sin
cos	cos
tg	tan
ctg	cot
sec	sec
cosec	csc
sh	sinh
ch	cosh
th	tanh
cth	coth
sch	sech
csch	csch
arc sin	$\sin^{-1}$
arc cos	$\cos^{-1}$
arc tg	$\tan^{-1}$
arc ctg	$\cot^{-1}$
arc sec	$\sec^{-1}$
arc cosec	$\csc^{-1}$
arc sh	$\sinh^{-1}$
arc ch	$\cosh^{-1}$
arc th	$\tanh^{-1}$
arc cth	$\coth^{-1}$
arc sch	sech^{-1}
arc csch	csch^{-1}

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rot	curl
lg	log

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RESULTS OF THE EXPERIMENTAL STUDY OF THE EFFECT ON COMPRESSOR PARAMETERS FROM WATER ADMITTED AT THE INLET TO A CENTRIFUGAL COMPRESSOR

A. S. Moskalenko, N. L. Zel'des

DESIGNATION LIST

воды = water

сух. возд. = dry air

влажное = wet

сухое = dry

In the case of an elemental gas jet moving in the rotor and adsorbing from the compressor impeller work dH, the equation of work between two infinitely close sections can be written in the form of the generalized Bernouilli equation:

$$dH = \frac{dP}{\gamma} + d\frac{C^2}{2g} + dZ + dH_r,$$

where P - pressure, γ - specific weight, C - absolute velocity, Z - height of arrangement, H_r - work on overcoming frictional forces.

It is evident that the work of friction dH_r is turned into heat and is imparted to the gas from within in the amount $dq_r = AdH_r$. Keeping in mind that with gases the change in potential energy of position dZ is infinitely small in comparison with other forms of energy of the jet, and multiplying the obtained equation by A = 1/427 (thermal equivalent of mechanical work), we obtain finally

$$AdH = AvdP + Ad\frac{C^2}{2g} + dq_r,$$

or, in finite form

$$AH = A\int_0^{\kappa} vdP + A\frac{C_\kappa^2 - C_0^2}{2g} + q_r. \tag{1}$$

The purpose of the compression process in a centrifugal compressor is to increase pressure. The power required to increase air pressure from P_0 to P_K, must be the least possible. Considering that velocities C_0 and C_K at the entrance and exit of the compressor are usually small and close to one another, we find from equation (1), using the theorem of mean value of the integral,

$$AH = A\int_0^\kappa v dP + q_r = Av_m (P_\kappa - P_0) + q_r,$$

or, substituting

$$v = \frac{RT}{P}.$$

(2)
$$AH = ART_m \ln \frac{P_\kappa}{P_0} + q_r.$$

These equations show that there are only two ways of decreasing the work of compression:

1) decrease the compression of mean specific volume v_m, in other words, decrease the mean gas temperature during compression T_m and

2) decrease the gas dynamic losses q_r.

Cooling the air compressed in the compressor by condensers and jacketed devices in elements of the compressor for circulating the

coolirg fluid must result in very considerable increase in the weight and dimensions of the motor and complicate its construction. Therefore, in aviation gas-turbine engines, if the compressed air is also cooled, then this is done usually only by the evaporation of fluid (water, alcohol, ammonia etc.) absorbed directly into the flow of compressed gas through special nozzles at the compessor entrance [1]. When liquid evaporates in compressed air, the latter gives off heat equal to the heat of evaporation. This decreases the polytropic index of compression, the work necessary to compress air to a given pressure, and also the temperature at the end of the compression. Evaporative cooling means are highly effective, and at the present are also used in turbocompressors. Esher-Wiess (Switzerland) has turned out more than 100 centrifugal compressors with evaporative cooling.

In 1967 the effect of water admitted at the entrance to a compressor on the operation of the compressor itself was studied experimentally on a stage of the D70-80-001 sb compressor ($\alpha_{3л} = 16°$; $Z_g = 24$; $\delta = 0.5$ mm).

Experimental Results

Experimental studies were conducted on compressor revolutions of n = 5200 r/min, n = 7100 r/min, n = 9000 r/min, n = 11,000 r/min, n = 12,500 r/min, n = 14,000, n = 15,500 r/min with different positions of the butterfly valve.

The flow rate of water varies from 10 g/s to 80 g/s.

The first experiments encountered difficulties caused by the drops hitting the thermocouples at the compressor entrance.

Computations of static temperatures in the smallest section of the measuring device, at the compressor entrance, and after the compressor, conducted from the tables of gas-dynamic functions [2], showed that the static temperature on the compressor entrance differs from the total temperature by not more than 0.1% Therefore a protective deflector was placed on one of the two thermocouples.

When judging the results of the experimental study, it must be kept in mind that the injection of water occurred in a compressor designed for dry compression.

Degree of Pressure Increase

The main advantage of water injection on the compressor entrance is the increase of achievable pressure increase. The effect resulting in pressure increase is similar to the effect from the supply of cooled air to the compressor entrance when the degree of pressure increase with given available work becomes larger.

Sometimes the pressure is increased in a wind tunnel which has no mechanical equipment by means of cooling due to the evaporation of liquid in air flow [3].

When water is injected into a flow of air moving along a channel of constant cross section without energy supply from without, it is possible to separate three conditions [4]. In the first set of conditions the speed of air relative to drop is large. Therefore the effect of channel resistance predominates, and stagnation pressure decreases in spite of the fact that the rate of vaporization is maximum. In that time the drops become heated and very rapidly achieve temperatures close to the dew point. At such a temperature the delivered heat goes entirely to vaporization. After achieving the dew point, the temperature of a drop remains practically constant.

In the second set of conditions the relative speed, and

consequently also the resistance, become very small, and vaporization of the drops predominates. Total pressure now increases, and becomes greater than its initial value. In view of the decrease in air velocity in this set of conditions, after a period of acceleration the drops achieve the speed of air, after which they move more rapidly than the gas and the corresponding resistance becomes negative. This must accompany an increase in total pressure. However, the noted effect is exceptionally small since the speed of the drop remains very close to that of the air.

In the third set of conditions the air temperature, as a result of continuous decrease becomes sufficiently close to the temperature of the drops, and the drop diameter is so small that the rate of evaporation becomes very low. Wall friction now becomes the predominant effect, and the amount of total pressure achieving a certain maximum value again begins to decrease.

One of the most important parameters is the initial diameter of the drop. When it changes, the length of the channel necessary to vaporize a given amount of liquid changes approximately in proportion to the square of the diameter [4].

For extremely large drop diameters the rate of vaporization can be so small that the second set of conditions completely disappears.

Figure 1 graphs the change in relative degree of pressure increase (ratio of degree of pressure increase with injection of water to the degree of pressure increase without water injection) as a function of the relative flow rate of the water $\left(\frac{G_{\text{возд}}}{G_{\text{сух. возд}}} = \bar{G}\right)$. The maximum degree of pressure increase with water injection for a given set of compressor operating conditions when n = 15,500 r/min, Z = 7/16 (Z - position of butterfly valve), is achieved at G = 0.026 [kg water/kg dry air]. The degree of pressure increase is 2.3% greater than the degree of pressure increase without water injection. Up to $\bar{G}$ = 0.026 the relative degree of pressure increase $\left(\frac{\pi^{*}_{\text{к влажное}}}{\pi^{*}_{\text{к сухое}}}\right)$ rises continuously. In this case up to $\bar{G} \cong 0.01$ the relative degree of pressure increase rises insignificantly, which is caused by the low flow rate of the water and by the low vaporization rate, since the nozzle is operating on small differentials at such a flow rate. With a slow rate $\bar{G} > 0.014$ the rate of increase $\frac{\pi^{*}_{\text{к влажное}}}{\pi^{*}_{\text{к сухое}}}$ is considerably greater. Achieving maximum, $\frac{\pi^{*}_{\text{к влажное}}}{\pi^{*}_{\text{к сухое}}}$ begins to decrease at $\bar{G} > 0.026$. This is explained by the fact that when G = 0.026 the moist air is in the region of saturation and further increase in the flow rate of the water does not cause a temperature decrease, i.e., at density increase, but only results in additional energy losses, frictional losses between the drops of water and air, between drops of water and channel walls, impact losses due to drops of water striking the

blades.

Flow Rate of Dry Air

The change in flow rate of dry air is graphed on Fig. 2.

When the amount of water fed into the flow increases, the mass flow per second of dry air rises continuously. This is explained by the increase in air density due to the pressure increase and also due to the temperature drop. Up to a relative water flow rate $\bar{G} = 0.014$ the flow rate of air rises somewhat more slowly than on section $\bar{G} = 0.014$-0.025. The maximum flow rate of dry air occurs at $\bar{G} = 0.025$. In this set of conditions the flow rate of dry air is 4.8% greater than the flow rate of dry air without injection of water. With a subsequent increase in the flow rate of water, i.e., when $\bar{G} > 0.025$, the flow rate of dry air begins to drop. This is explained by the fact that the tempera ure after the compressor remains virtually constant, but pressure decreases due to the additional energy loss caused by the presence of unvaporized water particles in the flow.

Air Temperature Change After Compressor

The change in temperature on the compressor $(T_k^{\bullet} - T_0^{\bullet})$ is graphed as a function of the water flow rate ($\bar{G}$) on Fig. 3, and characterizes the rate of water evaporation. On section $\bar{G}$ = 0-0.01 the degree of temperature decrease is somewhat less than on section $\bar{G}$ = 0.01-0.0185. The reason for this is that when the water flow rate is low, the differential on the nozzle is small, the quality of spray is bad and therefore the rate of evaporation is lower.

When the flow rate of water increases, the spray becomes finer and the rate of evaporation increases. When the flow rate of the water excedes 0.0185 the relative moisture content approaches unity; therefore the degree of pressure drop decreases. With a flow rate of $\bar{G}$ = 0.034, temperature is virtually equal to the temperature at $\bar{G}$ = 0.025. This indicates that a state of saturation is setting in.

Dependence of Power Required by Compressor Upon Flow Rate of Water

An increase in the flow rate of water means that the power required by the compressor per 1 kg dry air decreases continuously;

this is a consequence of the decrease in mean specific volume during compression, in other words, a decrease in the mean air temperature during compression (Fig. 4).

When the flow rate of water increases, a different law governing change in power is observed. Change in flow rate of water from 0 to 0.014 results in a decrease in required power by 3.2% and a decrease in relative water flow rate from 0.014 to 0.025 results in a power drop by 10.8%, which is explained by the higher quality of liquid spray at increased flow rates. In conditions with a water flow rate of 0.025 power changes insignificantly.

Decrease in power with water injection amounts to 15% The obtained results show good agreement with values introduced by V. F. Ris [5].

Percent of Water Evaporation

The relative content of water at the temperature exit is determined by the condensation method.

The results of processing experimental data indicate that prior

to the state of saturation in the compressor the entire amount of water fed through the nozzle evaporates. This is ensured by the good atomization of the nozzle, the characteristics of which were taken on a special installation.

The effect of a differential on the nozzle on the degree of fineness of atomization was studied. The results of the study indicate that an increase in pressure differential on the nozzle causes the diameter of the drop to decrease, at first rapidly, then more slowly.

Total evaporation of water particles in the compressor is also indicated by the amount of vaporized water as obtained from the equation of energy balance written for the "entrance-exit" section of the compressor.

The temperature values fixed at a certain distance from the section in which temperature is measured after the compressor, also indicate total evaporation of water in the compressor. These values differ from temperatures recorded immediately on exit from the compressor in the amount Δt, which considers heat exchange on the "section K-control section" segment. Moreover, during moist compression Δt is virtually equal to Δt in dry compression.

Conclusions

1. Injection of water on suction into a compressor for the purpose of evaporative cooling of air considerably decreases the temperature of the latter on exit from the compressor.

Thus, with a water flow rate on the order of 0.025 the air temperature on the compressor exit decreases by approximately 43°C with an ambient temperature of 11.8-13.1°C and relative ambient humidity ϕ = 41-42%.

2. During compressor tests total evaporation of the injected moisture occurred to the state of saturation with different turns and different positions of the butterfly valve.

3. Evaporative cooling of air during compression in a centrifugal compressor operating at n = const decreases the power required by the compressor.

4. Evaporative cooling increases the degree of pressure rise and increases the flow rate of the air.

5. The introduction of evaporative cooling causes the characteristic curves of the compressor (Fig. 5) to shift somewhat into the region of high flow rate in connection with the increase in air density on the exit from the impeller.

6. It is necessary to study the effect of the positions of the water nozzle relative to the VNA intake edges on the compressor and the degree of fineness of atomization on the required power and efficiency of the compressor.

7. During the compressor operation (300 hours) no salt deposits were observed on the rotor or the blade diffuser.

8. Water was observed in the oil line of the lubrication system during the experiment. This indicates that the use of evaporative means of cooling requires structural measures to prevent water from getting into the oil.

BIBLIOGRAPHY

1. Реактивные двигатели. Пер. с англ. под ред. Н. Г. Дубравского. Оборонгиз, 1962.
2. Таблицы газодинамических функций. Институт им. П. И. Баранова, Изд-во Министерства авиационной промышленности СССР, 1956.
3. I. T. Hamrick, W. L. Belde. Some Investigations With wet Compression, Transactions of the ASME, v. 75, 1953, № 3.
4. Основы газовой динамики. Редактор Г. Эммонс. Пер. с англ. под. ред. Г. И. Баренблатта и Г. Г. Черного. Изд-во иностр. лит. М., 1963.
5. В. Ф. Рис. Центробежные компрессорные машины. Машгиз, М. — Л., 1951.

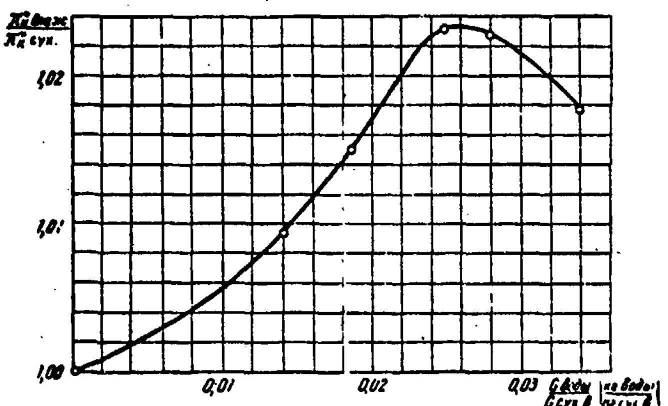

Fig. 1.

Fig. 1. Relative degree of pressure increase $\left(\frac{\pi^*_{к\ влажное}}{\pi^*_{к\ сухое}}\right)$ plotted against the flow rate of water $\left(\frac{G_{воды}}{G_{сух.\ возд.}}\right)$ when n = 15,500 r/min and Z = 7/16.

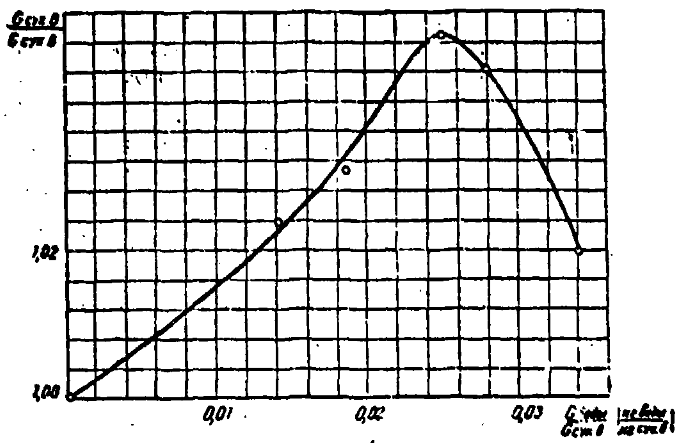

Fig. 2. Flow rate of dry air plotted against water flow rate when n = 15,500 r/min and Z = 7/16 ($G^{1}_{сух.\ возд.}$ - flow rate of dry air with injection of water).

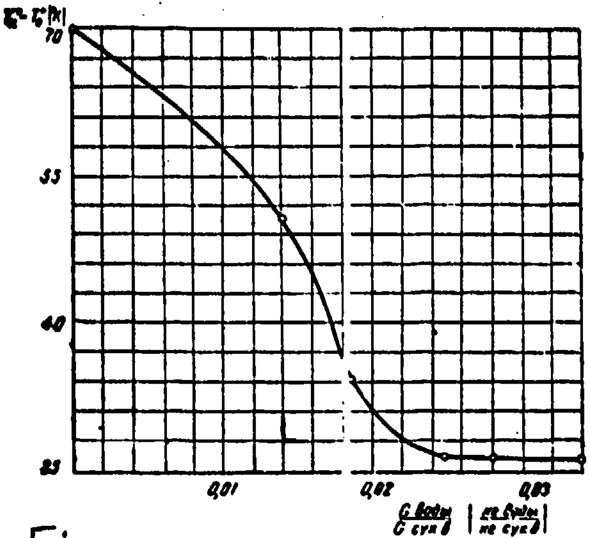

Fig 3.

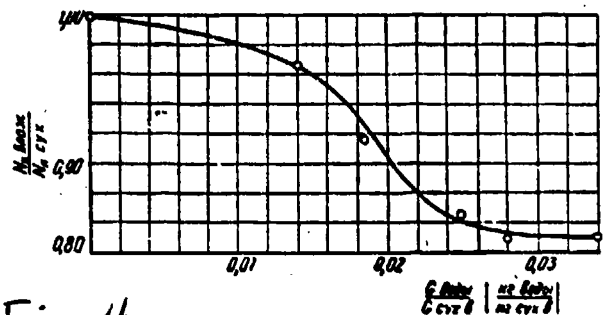

Fig. 4.

$-\circ-\circ-\ \frac{G_{\text{воды}}}{G_{\text{сух. возд.}}} = 0;$

$-\Delta-\Delta-\ \frac{G_{\text{воды}}}{G_{\text{сух. возд.}}} = 0{,}010 \left[\frac{\text{кг воды}}{\text{кг сух. возд.}}\right];$

$-\square-\square-\ \frac{G_{\text{воды}}}{G_{\text{сух. возд.}}} = 0{,}015 \left[\frac{\text{кг воды}}{\text{кг сух. возд.}}\right];$

$--\circ--\circ-\ \frac{G_{\text{воды}}}{G_{\text{сух. возд.}}} = 0{,}020 \left[\frac{\text{кг воды}}{\text{кг сух. возд.}}\right];$

$+\Delta+\Delta+\ \frac{G_{\text{воды}}}{G_{\text{сух. возд.}}} = 0{,}025 \left[\frac{\text{кг воды}}{\text{кг сух. возд.}}\right].$

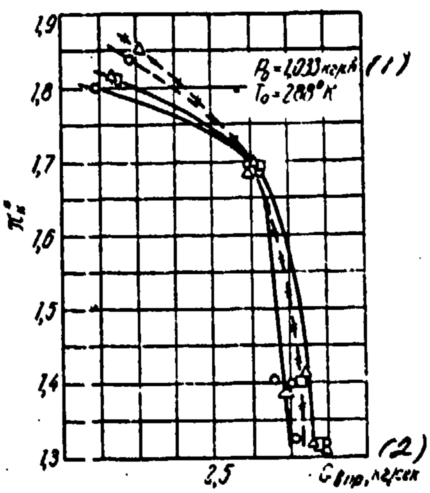

Fig. 5.

Fig. 3. Change of temperature in compressor as a function of water flow when n = 15,500 r/min and z = 7/16.

Fig. 4. Power required by compressor as a function of water flow rate at n = 15,500 r/min and Z = 7/16.

Fig. 5. Characteristic curves of compressor at n = 15,500 r/min and different water flow rate:

Key: (1) kg/cm^2, (2) kg/s.

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